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DESIGN FEATURES OF AGRICULTURAL MACHINERY FOR DEVELOPING COUNTRIES[1/]

by

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1 INTRODUCTION

One of the distinguishing features of developing countries is the importance of agriculture in their economies. Up to 90 per cent of the population may be employed in agriculture and most of the exports are agricultural products. In these conditions production of farm machinery for the home market appears a likely industrial development which could form a nucleus for developing skills in engineering design, manufacture and servicing.

Agricultural holdings can be broadly classified into three categories for the purposes of mechanization:-

(i) Subsistence farming in which hand tools are widely used. There appears to be considerable scope for a reduction in the number of designs of handtools. With standardization there are economies in large scale production. However, the developing countries are unlikely to be competitive in the manufacture of simple hoes because of the high cost of imported steel and the possibility of automating the forming of the tool so that the labour content is negligible. The main engineering requirements in this category are concerned with community development viz. water supply, roads, buildings.

(ii) Small progressive farmers who possess rudimentary standards of farm management. The source of power may be animal or small tractors. Considerable progress has been made in the development of animal drawn equipment and it has had a stimulating effect on farming practices. It is probable that in many areas animals are transitional between hand tools and engines. A better understanding of the principle of action of the implement and its place in the production system should help to establish implements well suited to local conditions and worthy of local manufacture using a proportion of imported components.

(iii) Large scale cash farming which uses full size conventional tractors. In most cases there are opportunities for either modifying existing equipment or for developing new equipment in order to satisfy more fully local needs.

2. THE SOIL-CROP SYSTEM

Agricultural machinery provides considerable scope for the designer to produce features which exploit the opportunities offered by local agricultural conditions. Mechanical design is one of the design problems of farm machinery. A more important problem is to decide how best to relate the specification for the machine to the whole system of which it forms a part. The machine is one link in an agricultural production system and the performance of the machine within the overall system is more decisive than the way it functions in the sub-system in which it operates.

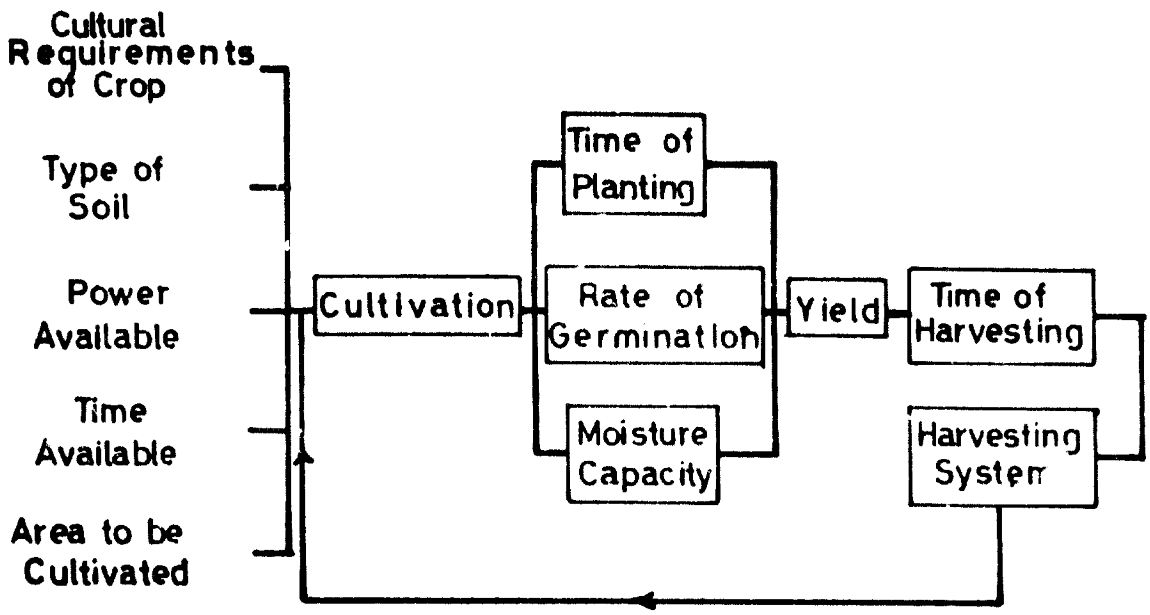

Fig. 1 Soil - Crop System

Some of the factors which help to specify the system performance of a cultivator are shown in Fig.1. The function of a cultivator may be described as the creation of a satisfactory environment in the soil for the development of the crop. In temperate climate conditions, the requirements for weed control and a level seed bed are decisive in selecting a mould-board plough as the primary tillage implement in the majority of cases. In a tropical climate, the need to plant before rain, to improve infiltration rates and to control erosion poses a different situation from that in temperate climates and another type of primary cultivator than the plough may be a better choice.

Some of the factors which influence the general capacity of the cultivator are shown on the left of Fig.1. The cultural requirements of the crop may lay down the need for a certain depth of cultivation or arrangement of rows. Usually there are wide limits within which crop growth is possible and unless there is evidence of the failure of a method it need not be ruled out as impossible.

The strength of the soil determines the force required to rupture it with a tined implement, disc, or mouldboard plough. If the designer can specify the local soil conditions, he is able to make the calculations for an economical mechanical design and for matching the implement to the source which is available to power the implement.

The capacity of the power source which is available to pull the cultivator decides the width of cultivator which may be used in a particular soil. By combining the width of cultivator with the time which is available for carrying out the cultivation it is possible to calculate the area for which one unit is suitable.

On the right of Fig.1 are shown some of the factors in the production of the crop which are influenced by the cultivation. The time of planting is related to the number of pre-planting cultivations which are necessary and the time required to carry them out. It has been demonstrated in several trials that crop yield is very dependent on the time of planting. The type of cultivator which is selected must give the farmer reasonable control over planting dates.

The percentage germination depends on the environment created by the cultivator in the soil. If there is a difference in the germination of seeds between two types of cultivator obviously the loss due to this factor must be taken into account in assessing the performance of the two implements.

The rate of infiltration of moisture into the soil during rain and the rate of evaporation during dry weather are very dependent on the action of the cultivator. Where water is a factor limiting crop yield, the relationship between type of cultivation and water storage is critical.

The time of planting, rate of germination and amount of water storage influence the yield of the crop and the time of harvest. Indirectly it is possible to link the final yield and to some extent influence the time of harvest by the capacity and type of cultivator which is chosen.

The method of harvesting may in some cases impose limitations on the kind of cultivation which should be carried out. For example the harvester may require level ground conditions for a cutting mechanism or absence of clods for a soil separating mechanism.

The systems approach to design (1) which has been outlined above uses as the design criterion the satisfactory operation of the total project rather than the independent consideration of the efficiency of the component parts of the system. The attention of the designer is directed to the complete project and he is provided with a framework for examining it. It is the approach of the agricultural engineer to farm machine design because he is able to look at both the agricultural production process and the mechanical design of the components in a similar quantitative way. The systems approach to design directs the attention of the designer to the total project which is complimentary to that of mechanical design, which directs his attention towards an analysis of the component parts of a machine.

2.1. The Design Criterion

In any systems approach to design it is necessary to compare different arrangements of sub-systems in order to reach a decision. The most practical common denominator is an economic comparison of different solutions. The criterion of selection can be either the lowest cost solution or the maximum profit solution.

By means of economic comparisons of different agricultural production systems it is possible to decide between hand labour and power machinery systems. Even if the hand labour system appears cheap in terms of capital investment, it incurs penalties in terms of the losses which arise through inability to plant at the correct time, low water storage and late harvest.

Unfortunately it is not possible in most cases to cost different systems with any accuracy. However, since comparative costings offer the only basis of long term decision making it may be necessary to start with estimates which can be improved as data are collected.

One example of the systems approach to machinery costing is the modification of the harvesting cost of a combine harvester by taking into account losses in yield due to harvesting at times either before or after the point of optimum maturity.

2.2 Machine Charges

Agricultural machinery is usually costed on a fixed cost-running cost basis. The fixed costs include depreciation, interest on capital, tax, insurance and storage. In calculating depreciation on combine harvesters a straight line method is used, allowing a scrap value of 10% of the initial price. The working life of a combine has been estimated in two ways:-

(i) 8 year life where the working load is less than 200 hours per season. In this case the useful life is terminated by obsolesence.

(ii) 5 year life where the working load is more than 200 hours per season.

As a result of this assumption, two costs may be calculated for a working load of 200 hours; the mean value was used in drawing Fig. 2.

In conjunction with the straight line method of calculating depreciation, interest at 8% per annum is charged on half the capital, the average investment in the machine.

The running costs are fuel, repairs and renewals, labour and charges for a tractor in the case of the trailed combine. The hourly scale of running costs is shown in Table 1.

Table 1. Scale of variable charges

Repairs and renewals	0.0003 (purchase price) per hour
Fuel - lubricants	3/3d. per hour
Labour	7/-d. " "
Tractor	7/-d. " "

The annual machine costs have been calculated for three different classes of combine harvester - trailed; medium size, self-propelled and large size, self-propelled.

Type	Capital Cost	Field Capacity
Trailed	£1,000	1 acre per hour
Medium Self-propelled	£2,400	2 " " "
Large " "	£3,500	2.5 " " "

Table 2 Details of Combine Harvesters.

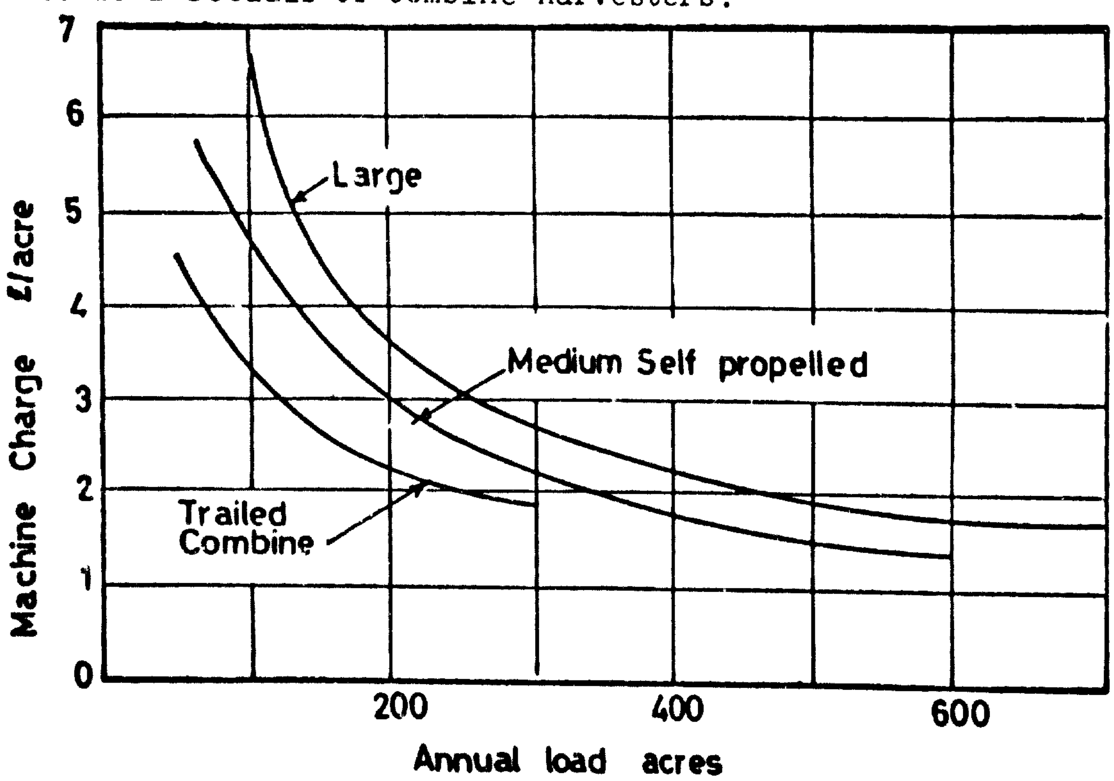

Fig.2 Machine charges as a function of annual load

All the curves for machine charges are of hyberbolic shape - the machine charge falls rapidly as the annual load factor increases and the charge reaches a relatively constant value at high load factors. However when the complete harvesting system is examined, it becomes obvious that there are other costs being incurred within the system which more than cancel out the reductions in machine charges which are brought about by a high annual load factor. One such important modifying factor is the effect of timeliness in harvesting.

2.3. Timeliness

One of the characteristic features of agricultural products is that the yield and quality of the product are dependant on the time of harvest. For some fruits the time of harvest may be as short as one day. For cereal grains the optimum period of harvest is from seven to fourteen days. If the crop is cut too early there is loss in yield due to incomplete development of the grains and loss in quality due to a high proportion of immature grains. When the crop has ripened losses occur due to shedding. Studies on losses in winter and spring wheats have shown that the losses vary from one variety to another and are influenced very greatly by the weather conditions at the time of harvest.

The machine cost of harvesting decreases as the load factor of the combine harvester increases in the manner shown in Fig. 2. However as the load factor increases, the probability of harvesting the grain in the optimum condition decreases. One way of taking account of timeliness is to make additional charges for the losses from the crop when harvesting takes place outside the optimum period. A linear scale of charges at the rate of ½% loss per day is shown as a function of timeliness in Fig. 3.

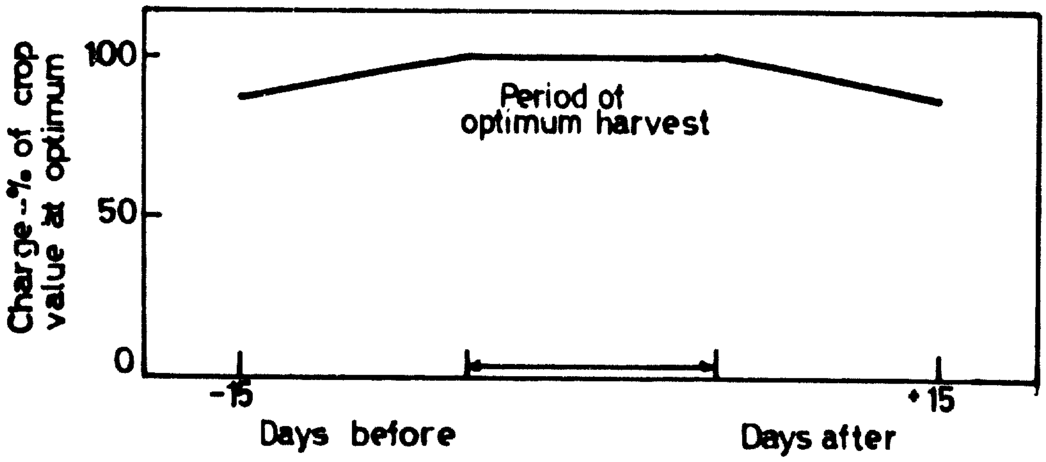

Fig.3 Effects of timeliness on charges in harvesting cereals

The following example shows the effect of timeliness on the cost of harvesting a crop of wheat under British conditions:-

Days available for harvesting during which a succession of crops is in optimum condition	= 20
Hours per day during which harvesting is possible at optimum moisture content	= 10
Value of crop per acre	= £40

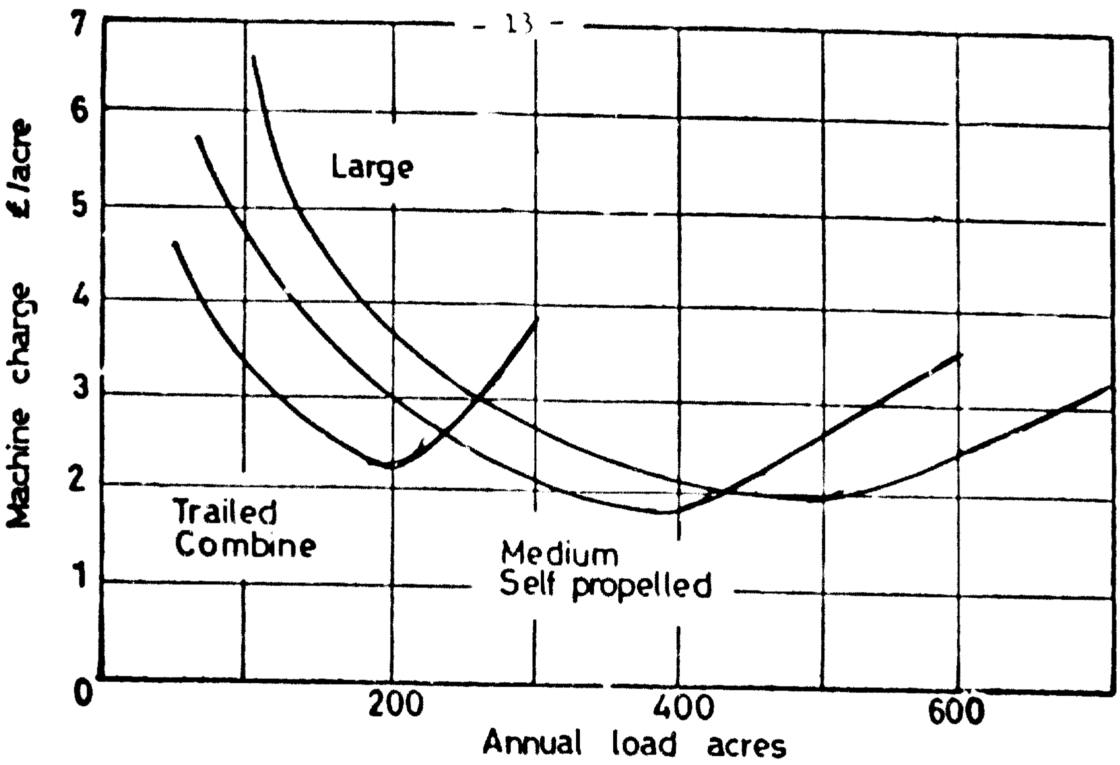

Fig.4 Effects of timeliness on machine charges.

The small trailed combine harvester which has a harvesting capacity of 1 acre per hour is capable of handling 200 acres before the farmer suffers loss due to shedding of the crop. Assuming that the loss increases at the rate of half per cent per day the cost of harvesting increases very rapidly as shown in Fig. 4. The medium self-propelled combine is capable of harvesting 400 acres and the large self-propelled machine is capable of harvesting 500 acres in the optimum condition. An additional advantage of the larger machine is that the rate of increase in the penalty for loss of timeliness is less than for the smaller machine.

Increasing the number of hours worked during each day may seem to be a possible way of extending the capacity of a small combine harvester so that more of the crop is harvested in the optimum condition. However it has been found that standing crops of cereals pass through a daily cycle of changes in moisture content. In Britain the moisture content reaches a maximum about midnight and a minimum in the early afternoon. As the moisture content increases the threshing losses increase. If the losses are allowed to reach a value of 5 per cent, the economic result is equivalent to doubling the machine charge per acre harvested.

Another effect of increasing the moisture content at which the grain is harvested is that it requires extra capacity for drying the grain and extra energy to removing the moisture. Drying charges may exceed harvesting charges.

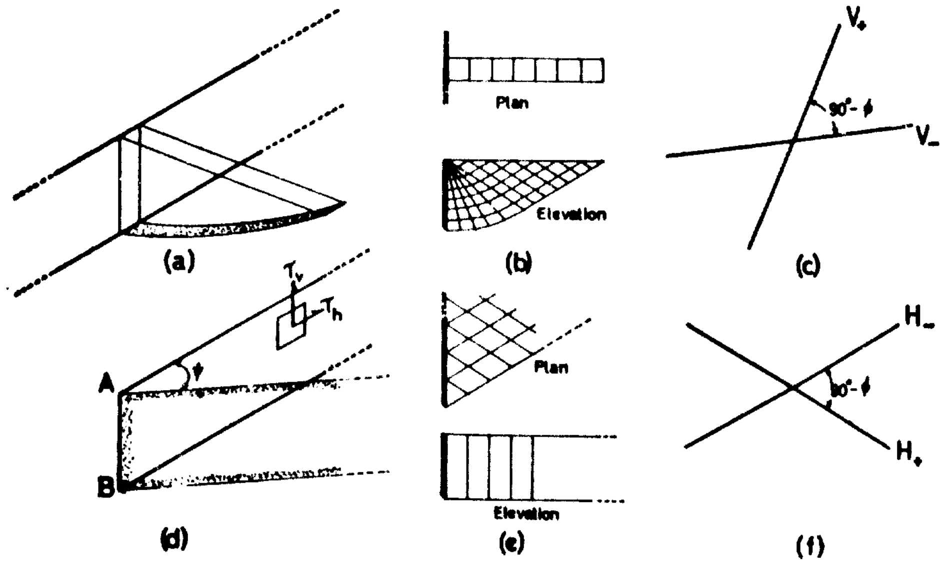

Fig.5 Vertical and horizontal failure regimes

3 CULTIVATION MACHINERY

Cultivation is usually the operation which consumes most power in the production of a crop. It is also the one which is most critical in influencing the yield from the crop through its timing and effect on water utilization. Decisions in regard to the width and type of cultivator which is to be used usually dictate the power source which will be needed.

3.1 Calculation of the forces on a tine

A large amount of research on two dimensional soil failure problems has been carried out in the Department of Agricultural Engineering, University of Newcastle upon Tyne. The solution of plane failure problems has been simplified by the use of an additive equation

$$P = \gamma Z^2 N_\gamma + cZN_c + C_a ZN_a + qZN_q \qquad .....(3.1.1)$$

where P is the passive soil force per unit width of the blade

Z = depth of blade

γ = soil bulk density

c = cohesion

C_a = soil-blade interface adhesion

q = surcharge pressure

N_γ, N_c, N_a, N_q are dimensionless numbers representing gravitational, cohesive, adhesive and surcharge factors.

Charts for these factors are available (2).

It is assumed that the soil is an incompressible rigid plastic material which fails according to the Mohr-Coulomb criterion. The shearing resistance of such a material is given by

$$\tau = c + \sigma \tan \phi$$

Hence the soil is described by only three variables - bulk density γ, cohesion c and angle of internal friction ϕ.

When a cultivator tine works in the soil, the end effects cannot be neglected. Hence the problem of soil rupture is not 2 dimensional but 3 dimensional. There is no rigorous solution for this problem but a semi-empirical solution based on the 2 dimensional case has been proposed by Hettiaratchi and Reece.(3)

An elemental strip within an infinitely wide vertical interface is shown in Fig.5 (a). The soil mass within the failure boundaries in front of this interface shows two planes of incipient failure in this 2 dimensional case. At any point within the soil mass these planes make an acute included angle of (90° - ϕ) with one another (c); the actual failure surface boundary will be composed of one of these planes. The slip directions labelled V+ and V- are contained in vertical planes and remain so for any inclination of the interface.

If the interface is finite, the edge AB forms a physical discontinuity, Fig. 5(d), and conditions similar to that obtaining at the bottom edge of the infinite interface would be expected to occur along AB. A further set of slip planes labelled H+ and H- Fig. 5 (f) will be associated with this case.

The basis of the analysis of the three dimensional soil failure is to assume that the failure configuration is composed of the two independent plane soil failure conditions corresponding to the vertical and horizontal conditions. The lower boundary of the finite interface ABCD shown in Fig. 6 will be identical to the elevation shown in Fig. 5(b). This failure surface CBEF (labelled (1) in Fig. 6) will intersect the two horizontal regime sliding surfaces (2) along the curves BE and CF. If the width AD (=2y) of the interface is kept constant and its depth Z is increased, the rupture distance X will increase until at some critical depth Z^1, the points E and F will coincide at the point G. The slip-planes in the horizontal regime will then form a complete wedge having an apical angle of $(90^o - \phi)$ and with its lower end truncated by the vertical failure regime Fig.6 (b). Any further increase in depth will cause the wedge sides to increase in depth as shown in Fig. 6 (c).

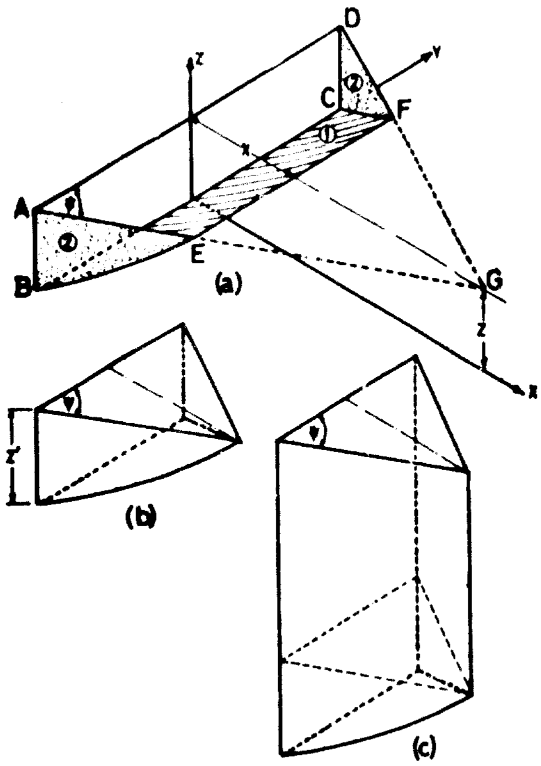

Fig.6 Evolution of three-dimensional soil wedge

The force acting on the tine has been shown (3) to be the vector sum of two components:-

(i) a forward failure force P_f which is associated with soil failure in vertical planes. It acts at an angle δ to the interface and its magnitude is given by eq. 3.1.1.

(ii) a corresponding sideways failure force P_s which acts normal to the interface is given by

$$P_s = \left[\gamma(d+\frac{q}{\gamma})^2 \; wN_{s\gamma} + cwdN_{sc}\right] K_a \qquad \ldots\ldots (3.1.2)$$

where $N_{s\gamma}$ and N_{sc} are dimensionless numbers representing cohesive and gravitational factors in the sideways direction. Charts of these factors are available (3).

The draft component D on a tine working at a forward rake angle α is

$$D = P_f \sin(\alpha + \delta) + P_s \sin\alpha + C_a Z \cot\alpha \qquad \ldots\ldots 3.1.3$$

and the vertical lift component L of the resultant force on the tine is

$$L = P_f \cos(\alpha + \delta) + P_s \cos\alpha + C_a Z. \qquad \ldots\ldots 3.1.4$$

A comparison of calculated and measured values of draft at three different settings of rake angle is shown in Fig. 7. The experiments were carried out in damp quarry sand with a tine 2 in. wide.

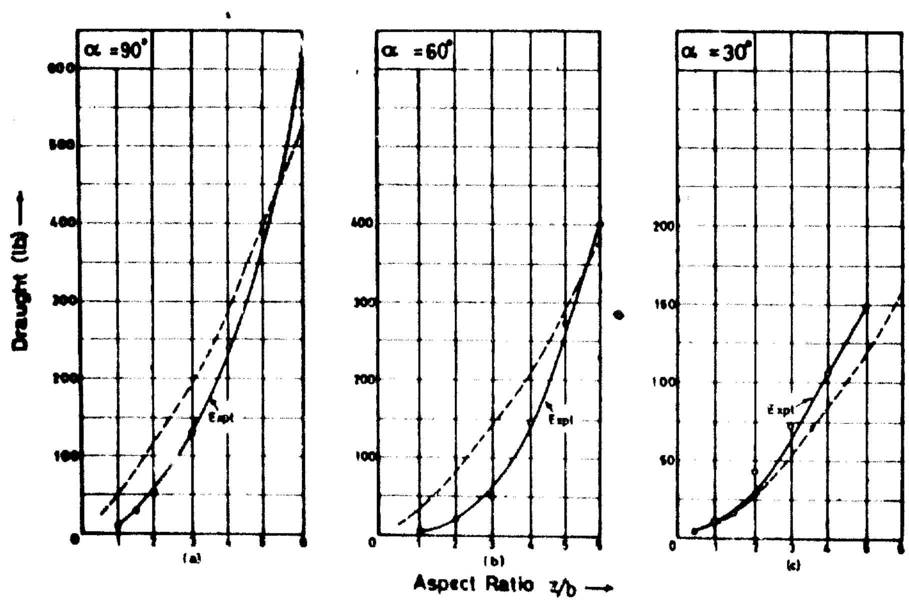

Fig.7 Draught forces of a narrow tine in damp quarry sand

The forwards and sideways rupture distances are shown in Fig. 8.

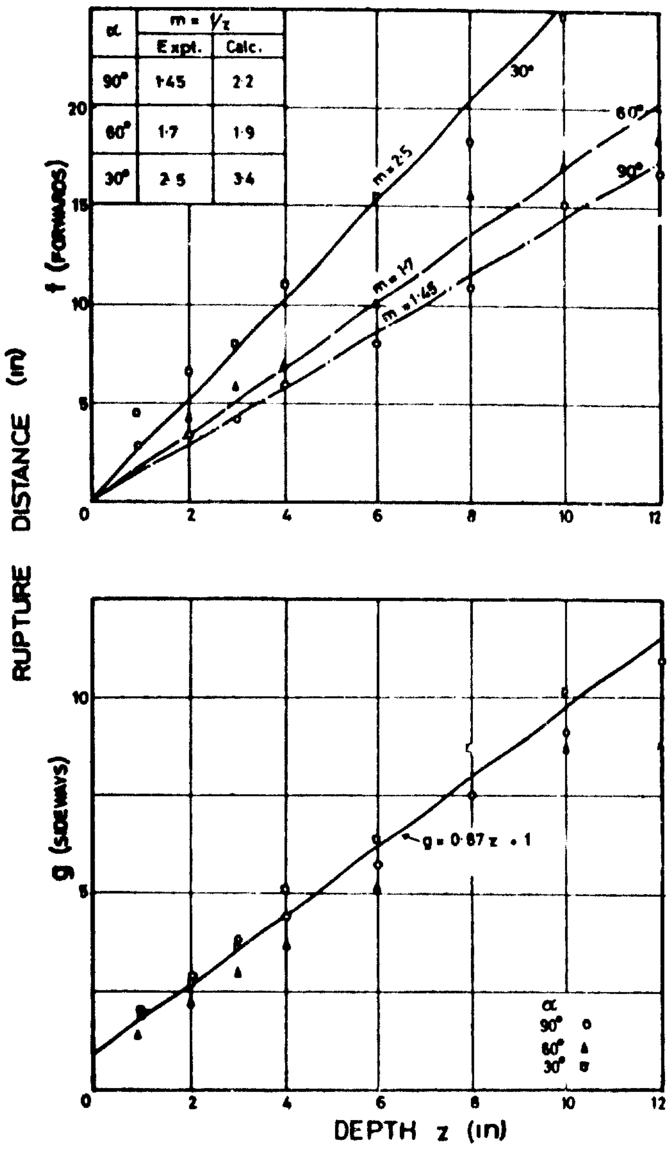

Fig.8 Forward and sideways rupture distances of a 2 in wide tine in damp sand.

Tined cultivators are capable of working to depths of 0.25m and single tines may be used for sub-soiling to depths of 1m. Tined cultivators are simple to design and manufacture. They appear to be very suitable for tropical conditions for the following reasons:-

(i) they do not remove plant residues completely from the surface nor produce a fine tilth which might be easily eroded away;

(ii) they are capable of working deeply in the soil which improves the permeability, reduces run-off at the surface and increases the amount of water which can be stored in the soil;

(iii) they are able to work in a wide range of conditions.

Small shallow-tined cultivators may be designed for use with animals. Full size tractors however are needed to provide the draft necessary to work deep tines.

3.2 Disc implements

Disc implements work the soil in a shallow layer near the surface. They have the advantage that most of the trash is retained near the surface. Disc implements will work rapidly in a wide variety of conditions. They produce greater pulverization than tined implements and they are not as effective in encouraging water infiltration as tined implements.

The forces on a disc 600 mm diameter have been analyzed by Johnston and Birtwistle (4) for Australian wheatland conditions. They examined the three perpendicular components of the force acting on the disc viz.

F_R the component of the force acting in the direction of motion;

F_S the component of the force acting perpendicular to the direction of motion, which forces the disc against the furrow wall;

F_V the component of the force acting vertically upwards and causing the disc to lift vertically out of the ground.

Three other important variables in the operation of a disc are:

β the drift angle which is the angle between the direction of motion and horizontal diameter of the disc;

θ the camber angle between the axis of the disc and the horizontal,

v the speed of forward travel of the implement.

The forces were expressed in terms of the cross-sectional area of the furrow, which was ploughed by the disc, and are shown as a function of drift angle at two values of camber angle in Fig. 9.

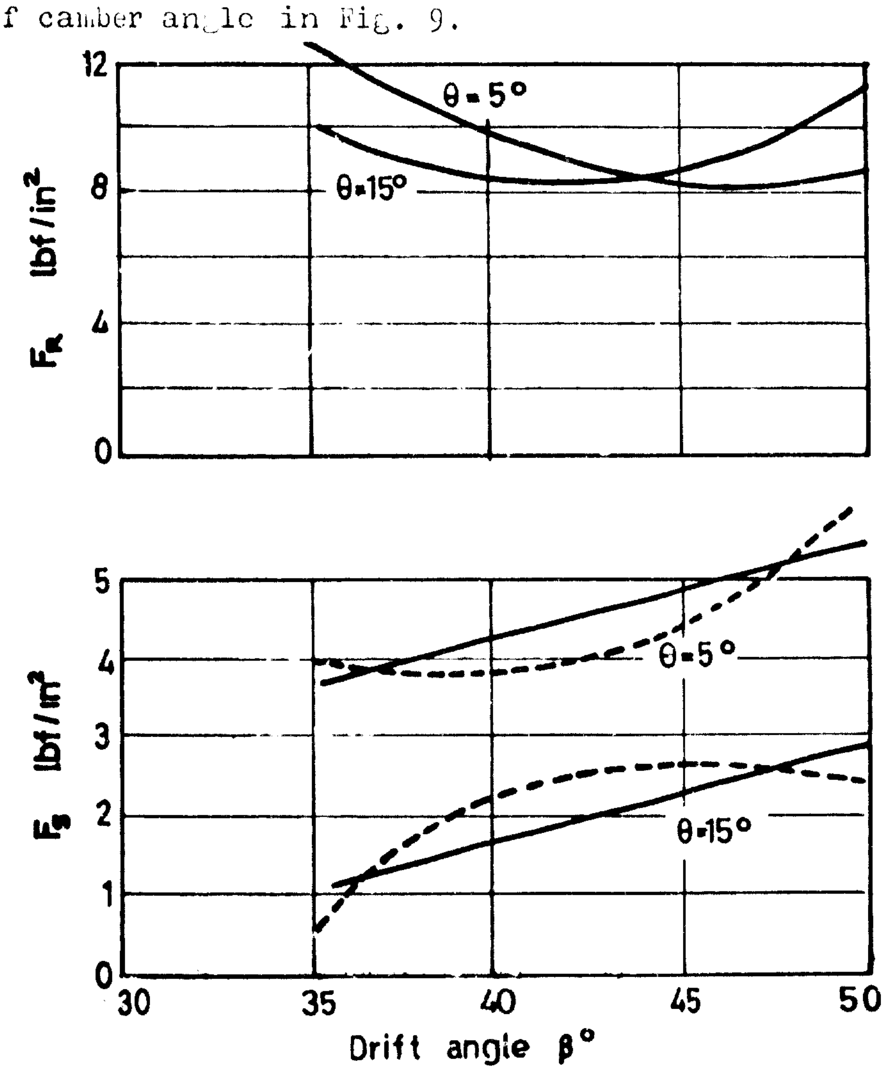

Fig.9 Forces on a disc plough working at 3½ in depth at approx. 2 mile / h.

Forward speed had a significant effect on the magnitude of F_R (as shown in Fig. 10). This effect has been reported by several workers.

The vertical force reached its minimum value at a drift angle of about 50 deg. F_V increased as the drift angle was either raised or lowered from 50 deg.

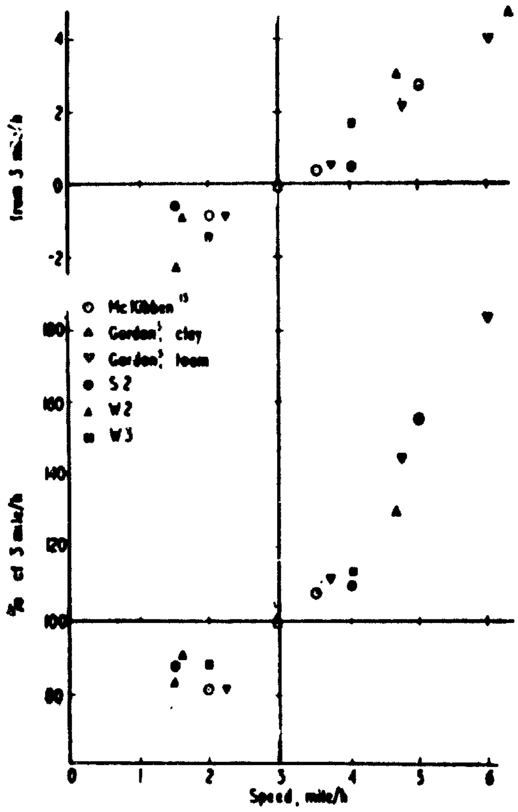

Fig.10. Increase of rearward force with speed

It is recommended that the full width of cut should be used with disc ploughs. This recommendation improves the working rate of the plough without a corresponding change in the draft per unit width but it does require extra weight on the disc to overcome the vertical force lifting the disc out of the ground.

The drift angle at which the draft force is a minimum varies with furrow width, soil type and camber angle. Increasing the camber angle caused a marked decrease in the side force.

3.3 The mouldboard plough

The mouldboard plough is unique among cultivating implements in the amount of weed control which is possible with it. The plough controls weeds by burying them as it inverts the furrow slices. The mouldboard plough is capable of working at depths from 100 - 300 mm, it exposes a completely new layer of soil and it increases the pore spaces in the furrow slices.

The action of the mouldboard plough is to separate a slice of soil by means of a horizontal and a vertical cut. The horizontal cut is made by the share whose action is an extreme case of a flat tine. The vertical cut is usually made by a flat disc. The mouldboard inverts the furrow slice by accelerating it, lifting it and displacing it sideways. The mechanics of handling soil by means of a mouldboard have been discussed by O'Callaghan and McCoy (5) who used a scratch-trace technique to find the path of the soil along the mouldboard. Curves were fitted to the scratch traces. The curves were differentiated and used to find the velocities and accelerations at different points along mouldboards of different shape. The work done by the mouldboard in accelerating the soil and against gravity, friction and adhesion were computed for a range of working speeds. The variations in draft with speed are shown in Fig.11.

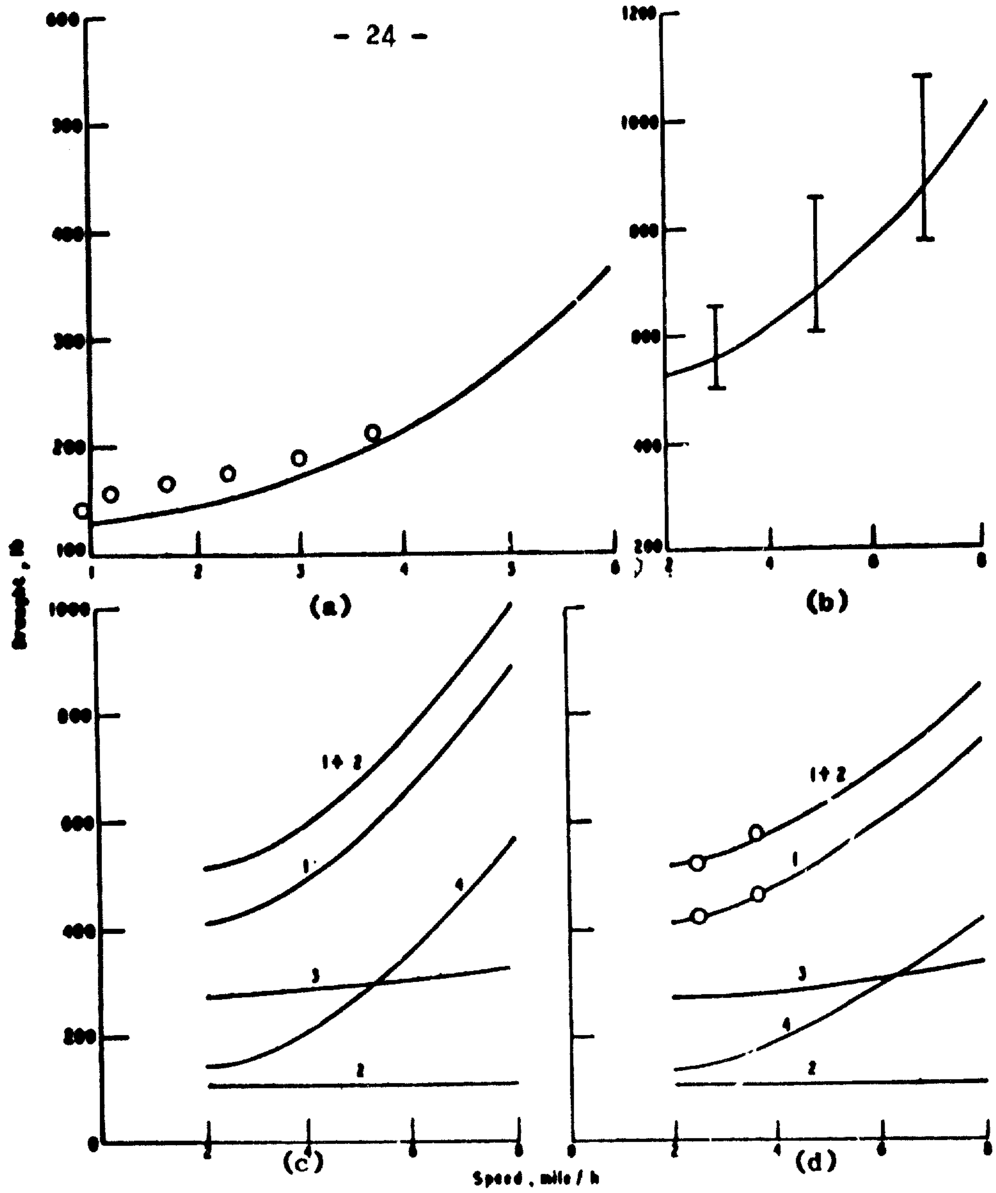

Fig.11 **Performance of mouldboard ploughs. Calculated and Measured values of (a) Mouldboard, (b) Plough. Draught due to (1) bodies,(2)coulters,(3)shares and (4) mouldboard of two types of ploughs (c) and (d).**

The analysis has shown that a mouldboard can only be designed to work effectively over a limited range of speeds. The acceleration which acts on the furrow slice depends on both the shape of the mouldboard and (velocity of travel)^2. The friction force depends on the acceleration. As the speed of working of tractors is increased, it is possible to maintain the acceleration applied by a mouldboard to the soil at a low value by changing the shape of the mouldboard.

3.4 The Mechanics of Root Growth

A laboratory study of the effects of mechanical constraints on the growth of roots in compact granular media has been carried out in the Department of Agricultural

Engineering, University of Newcastle upon Tyne (6). Barley and rice were grown in a modified cell of a triaxial shear testing machine for soil. The confining pressure surrounding the cell could be held at selected values while the growth of the roots was observed. Three different regimes of growth were identified:-

at cell confining pressures up to 3 lb/in^2 there was no effect on root growth, and the roots were of normal thickness;

at cell pressures over 8 lb/in^2 root growth was severely curtailed;

at cell pressures 3 - 8 lb/in^2 root growth was retarded and the roots thickened in diameter in an attempt to penetrate the medium.

The mechanism of root penetration in a confined medium appears to be one of radial thickening of the root cap which reduces the resistance of the soil to axial penetration followed by elongation. This cycle of thickening followed by elongation continues as the root grows downwards.

4 TRACTION

It is possible to calculate the draft required by each of the cultivation implements described above from a knowledge of the shear properties of the soil in which they are to work. In order to match these implements with a tractor an understanding of the variations in tractive performance under different soil conditions is helpful (7).

The shear stress developed by a flat plate which is in contact with the soil and is being moved through the soil is of the form shown in Fig. 12.

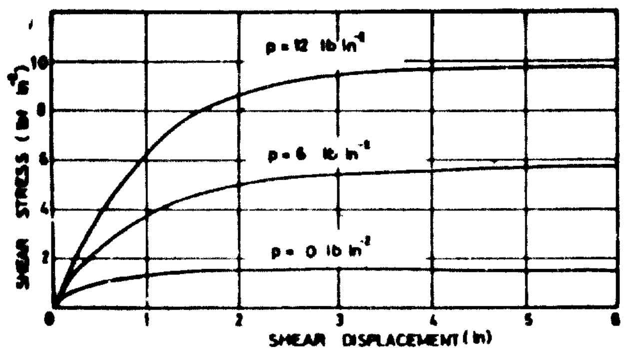

Fig.12 Typical shoar stress against shear displacement curves at three mean normal pressures, p.

The relationship between maximum shear stress and displacement is described by the equation

$$s = (c + \delta \tan \phi)(1 - e^{-j/k})$$

The Coulomb equation is modified by the deformation term $(1 - e^{-j/k})$.

where j is the linear soil displacement

and k is a shear displacement index.

Beneath a tractor wheel or track the shear deformation increases from the beginning of the wheel - ground contact where it is zero to the end of the wheel - ground contact where it is a maximum. If the relationship is assumed to be a linear function of x, the distance from the beginning of the wheel - ground contact area. The shear stress along the contact surface is given by

$$s = (c + \quad \tan \phi)(1 - e^{-ix/k})$$

where i is the slip, a dimensionless function.

Integrating this equation along the contact area gives the maximum thrust which can be developed by the wheel

$$H = (blc + V \tan \phi)\left(1 + \frac{k}{il} e^{-il/k} - \frac{k}{il}\right)$$

where b = width of contact area

l = length of contact area.

The 'slip-function' term $X = \left(1 + \frac{k}{il} e^{-il/k} - \frac{k}{il}\right)$ can be evaluated as a function of $\frac{il}{k}$ as shown in Fig. 13.

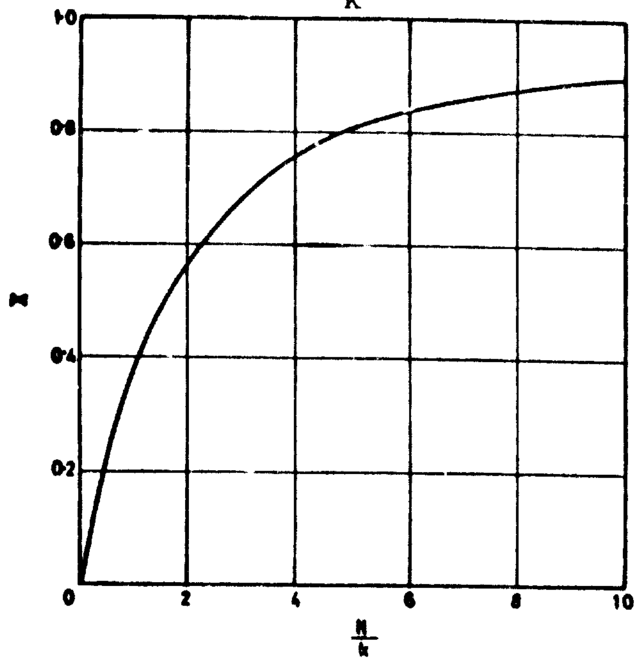

Fig.13 Variation of slip function X with ratio il/k

The maximum thrust $H = (blc + V\tan\phi)\,X$

The drawbar pull is found by subtracting the rolling resistance from the maximum thrust. If the rolling resistance is assumed to be constant = R

then maximum drawbar pull = $(blc + V\tan\phi)\,X - R$

and drawbar horsepower $P = \frac{S}{550}(1 - i)\left[(blc + V\tan\phi)\,X - R\right]$

where S is the speed of the tractor at zero slip

A typical curve of drawbar horsepower against slip is shown in Fig. 14.

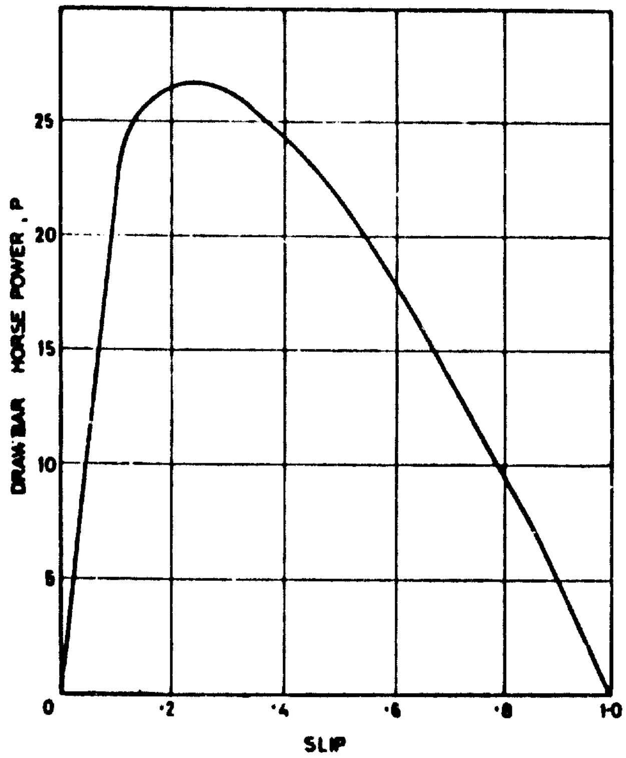

Fig. 14 A typical curve of drawbar hp against slip.

5 WATER SUPPLY

Water supply is one of the most widespread and uniform demands in rural areas. It can be seen as an aid to living conditions in rural communities, as one of the requirements of livestock and as a way of increasing production of arable crops by irrigation. The costs of pumping water are relatively low and simple water supply systems are convincing examples of the benefits of mechanization. The capacity of even small pumps is so great that communal ownership is possible without conflicting demands, especially if buffer storage is provided in the system. Such a water supply scheme can lead to co-operative ownership of other agricultural machines.

The most versatile pump for rural use is the jet-assisted, centrifugal pump. As an ordinary centrifugal pump it can be used for low head duties viz. lift 4 metres, delivery 10 metres. Applications are pumping drinking water from shallow wells or irrigation water from canals. The lift can be increased to 20 metres by by-passing some of the water from the delivery side of the pump back to a jet-injector, which is placed in the well on the suction pipe of the pump. In this way the pump and driving motor can be kept on the surface where they are accessible for maintenance and there are no difficult requirements in regard to sealing.

The pump could be driven by a small air cooled petrol engine. A one horsepower engine would be capable of pumping about 100 litres per minute. The installation is shown diagramatically in Fig. 15. The buffer store helps to even out the fluctuating demand, and reduces the number of times it is necessary to start up the engine. It can also act as a reservoir from which supplies can be distributed to several points. The capacity of the buffer store should be at least 50 times the delivery rate of the pump per minute.

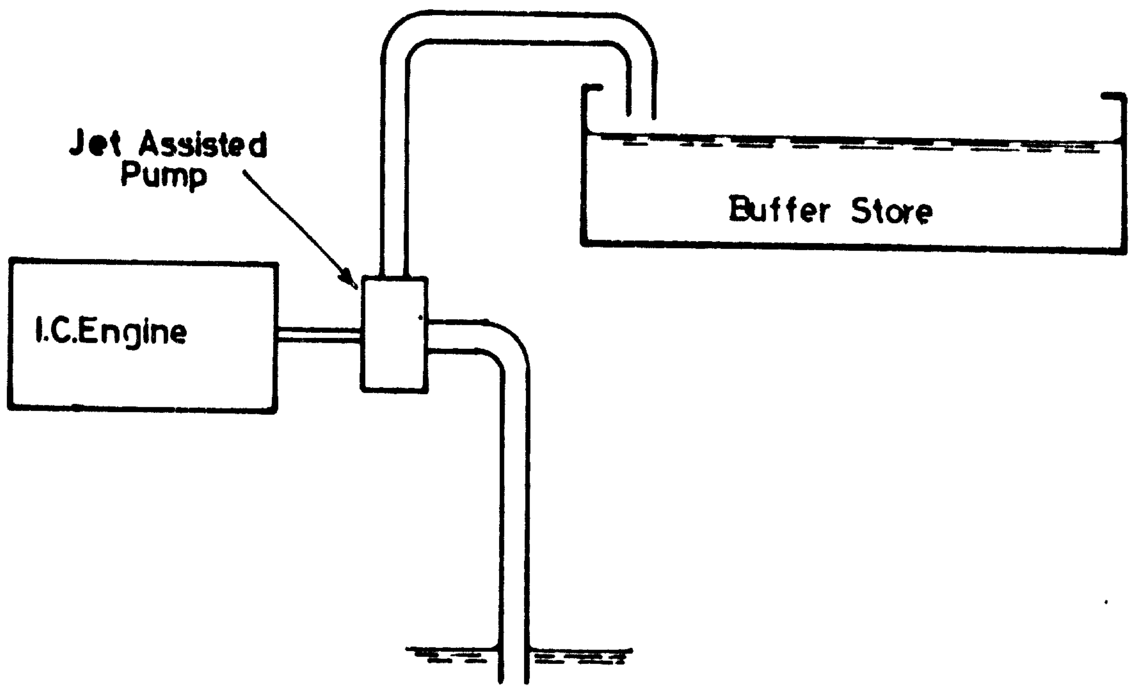

Fig.15 Simple pumped water supply

6 CROP STORAGE

One of the characteristic features of agricultural production is its seasonality. As a result most of the crop must be placed in storage as it is harvested and withdrawn to meet a demand which is usually uniform over the whole year. Most agricultural products require special treatment and careful storage if serious deterioration in quality and nutritional value is to be prevented. The main sources of loss are respiration, growth of bacteria and moulds and attack by rodents.

The principal methods of making plant materials safe for storage are drying and modification of the environment by refrigeration, lowering the PH, or removal of oxygen. The most widely used method is drying, mostly natural drying but also forced air drying under closely controlled conditions.

Since drying is a mass transfer process, the main design feature in a natural drying process is to create the maximum driving force. The controlling equation is of the form:-

$$\frac{\partial W}{\partial \theta} = kA\,(\rho_s - \rho_o)$$

where $\frac{\partial W}{\partial \theta}$ is the rate of drying

k is a mass transfer coefficient

A is the area through which moisture is transferred

ρ_s is the partial pressure of the moisture in the material to be dried

ρ_o is the partial pressure of the moisture in the air surrounding the material to be dried.

The main factor in the equation which can be influenced by the operator is ρ_o which has a minimum value corresponding to the relative humidity of the atmosphere. However within the bulk of the material it may rise to ρ_s unless a flow of air is maintained around the material which keeps the micro-climate in the same condition as the atmosphere. Flow of air is maintained through the material either by stacking it in thin windrows or by providing equipment which can turn the material at frequent intervals.

The simplest form of forced air dryer is the batch tray dryer which is cheap to construct and is capable of drying a very wide variety of materials. The minimum useful size is likely to be one based on a 5 horsepower fan.

When the moisture content has been reduced to a safe value for storage, the product must be made safe from rewetting, and attack by insects and rodents. Plastic materials have considerable possibilities for providing low cost protection in structures which allow the product to take up its natural angle of repose.

7 LAND CLEARING

An example of the design of a specialized tool for land clearing arose from the need for a small tractor mounted machine for bush clearing from small holdings in Uganda. Stumps from felled trees left after hand clearing are a major hazard to tractor cultivation. There is a long period each year when there is no agricultural work to be done and the tractors of the Government operated hire service are lying idle. Much of the agriculture in Uganda is of peasant type involving very small plots, often under 1 acre and rarely over 5 acres in size. Both the size of the holdings, and the cost involved mitigate against the use of heavy clearing equipment. The poor standard obtained by hand clearing usually necessitates the use of disc ploughs for subsequent cultivation. Where the clearing has been of a high standard, it has been possible to use mouldboard ploughs and to obtain complete coverage with good weed burial and an even surface after ploughing.

7.1 Problem Definition and Design Parameters

A device is needed which will remove small trees together with main roots down to a cultivable depth of 12 in. and which will be operated by hire service tractors on a contract basis. The unit will be required to operate in small holdings during the dry season and to have a high rate of tree removal, thus minimising cost per tree.

The radial limit of tree roots is not known although it is estimated that for the trees concerned the large roots will be confined to a 15 ft. radius central to the trunk. It is desirable that as much of the root growth as possible is removed although broken fragments left within the soil can be tolerated.

Recorded data is available concerning the horizontal forces required to push out or pull out trees. No data is available, however, on the vertical forces required to pluck a tree out of the ground.

It is thought that the proposed machine should be able to remove trees with trunk diameters ranging from 4" to 12". The cross section of tree trunks is irregular although it is found that the variation between the maximum and minimum diameters of a given profile are probably less than 25% of the mean diameter. It is suggested that contact between the machine and the tree should occur only up to a maximum height of 3 ft. from the ground due to a possible weakening of the trunk by termites above this limit.

The proposed machine will most probably be operated from a 50 h.p. tractor. The tractor operators, although trained tractor drivers, will have little mechanical knowledge and will be unskilled.

7.2 Methods of Tree Removal

Winching - The time taken to run out and attach the cable makes this a slow method, although by wrapping the cable around several trees so that each is pulled down in turn increases the speed of operation. This method is not generally favoured because of the associated dangers and safety hazards.

Pushing or Pulling - Pushing is to be preferred to pulling on the grounds of safety. Direct pushing is not possible due to lackof sufficient tractive force from the tractor. Sufficient force could be obtained by means of a hydraulic ram and a spragged anchor plate. As the point of application of the ram is raised above ground level the pushing force required will become less but the tree will be more liable to bend or break.

Plucking - This method envisages applying an upward force to the tree and lifting it bodily out of the ground with as much of the root growth as possible. Previous experience with this method has shown that the major problem is the application of the lifting force through suitable collars or tongs which grip the tree trunk. These have been found either to slip or roll up the tree or tended to crush the tree and sever it. Plucking the tree out in this way would seem to be a good method for removing much of the root growth with the trunk in one operation.

Sawing - Circular saws and chain saws are not acceptable on the grounds of safety. They also offer no possibility for removal or root growth.

Blasting - Not acceptable on the grounds of safety.

7.3 Design Solution

In the light of the defined problem and design parameters it was thought that a tree plucker offered the best solution if the practical difficulties already mentioned could be overcome.

Several design solutions based on this principle were proposed out of which the design for a prototype tree plucker emerged. Due to a lack of any factual data on the lifting forces required, the machine was designed to extert a maximum vertical pulling force of 40,000 lb at the tree trunk. This figure was estimated in part from a consideration of the shear strength of the trunk in contact with the gripping mechanism.

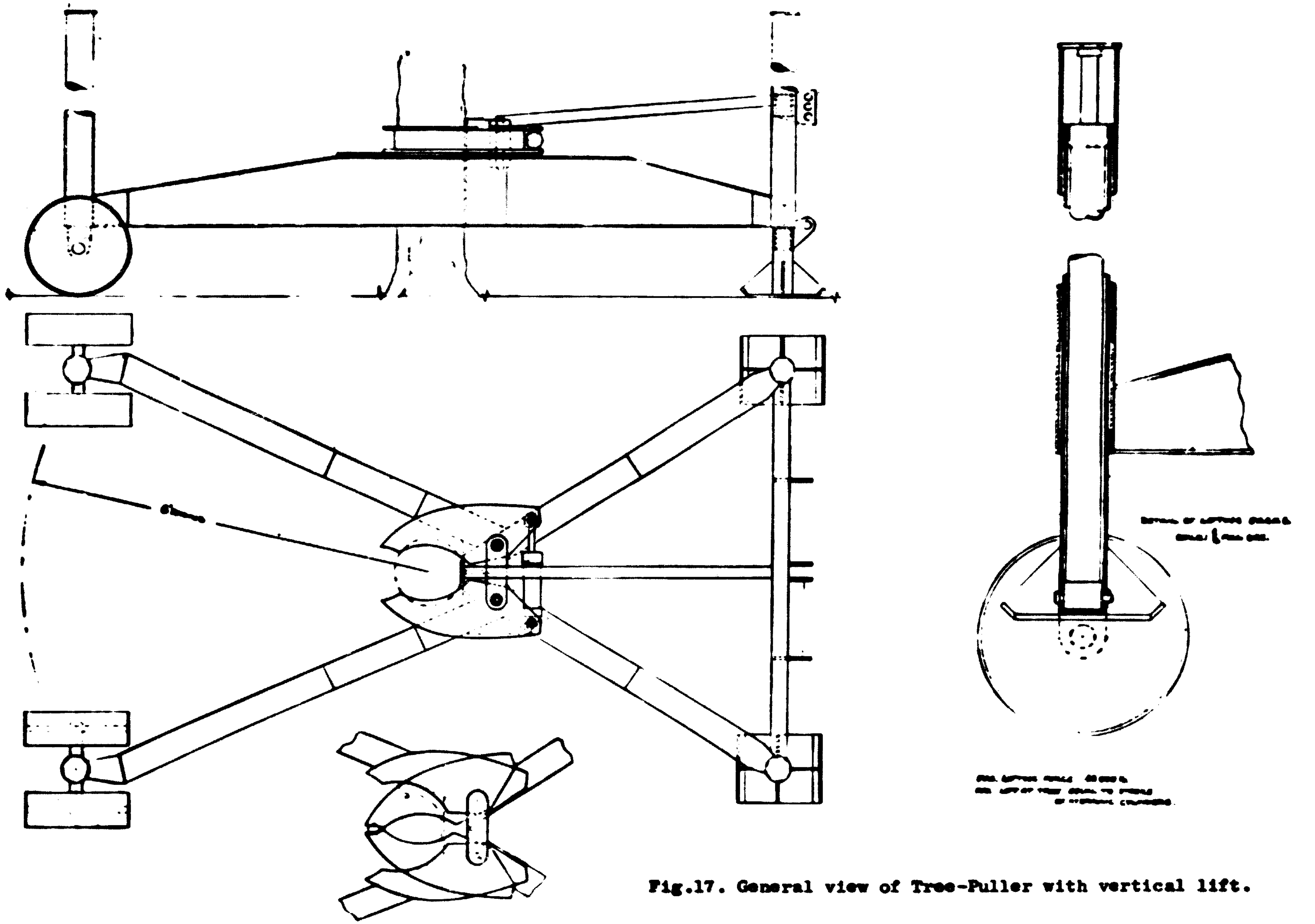

Fig.17. General view of Tree-Puller with vertical lift.

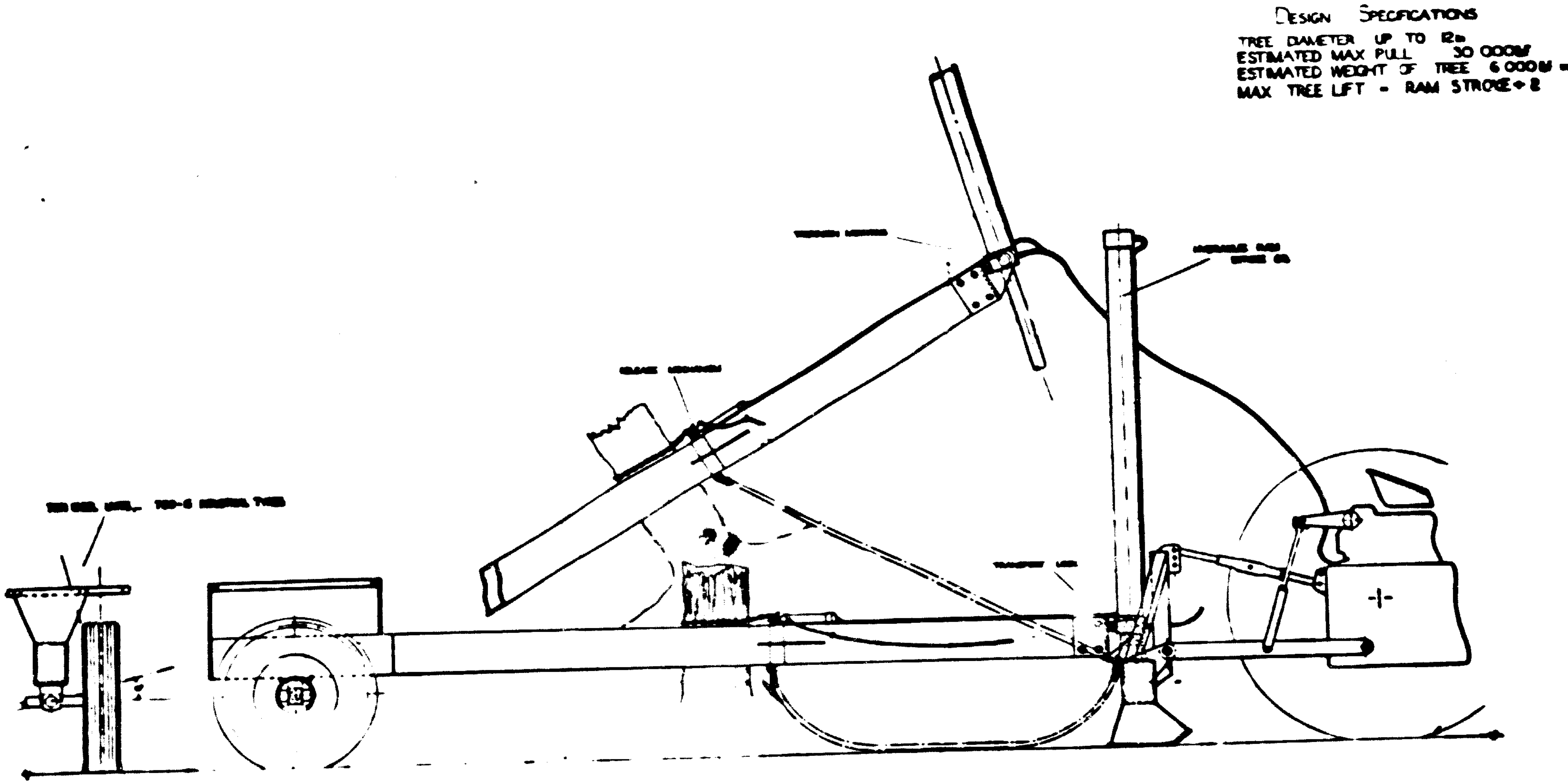

Fig. 16 General view of Tree-Puller with "hinged" lift arrangement.

The machine (Fig.16, 17) is essentially a tractor mounted loading frame open across the back end with four ground pads and a pivoted 'Y' shaped lifting frame. A large hydraulic ram between the front ends of the two frames provides the lifting force. Gripping jaws are mounted at the apex of the lifting frame. In operation, the unit is backed up to the tree, and lowered to the ground. The tree is then gripped and the upper frame pivoted by extending the ram, and thus lifting out the tree. The original gripping mechanism consisted of a chain tensioned hydraulically, but was replaced by hydraulically powered jaws which could be operated from the tractor seat.

The machine has been shipped to Uganda for field evaluation. Preliminary tests carried out in England showed that the machine is in fact capable of removing trees. Several orchard trees and a 16 in. diameter elm have been successfully removed. Root removal with the trunks has been most successful. The experience gained from these few field tests indicates that the machine can be made much lighter and that it will need to be much more manoeuvrable. Definite answers to these and further points such as the lifting forces involved and how close the machine can approach the tree trunk without impeding root extraction, should result from the field evaluation in Uganda.

8 DESIGN OF FRAMES

One of the main structural problems in many farm machines e.g. trailer bodies, cultivator frames, is the design of a framework which has a high degree of torsional strength and stiffness in relation to the material used in the frame. It has been shown (8) that diagonal bracing, either single or double, is a most effective way of increasing the torsional stiffness of a framework.

8.1 Analysis of force system in a framework

The deflected shape of the frame (bottom) and the force and moment system acting on two sides (right) are shown in Fig. 18.

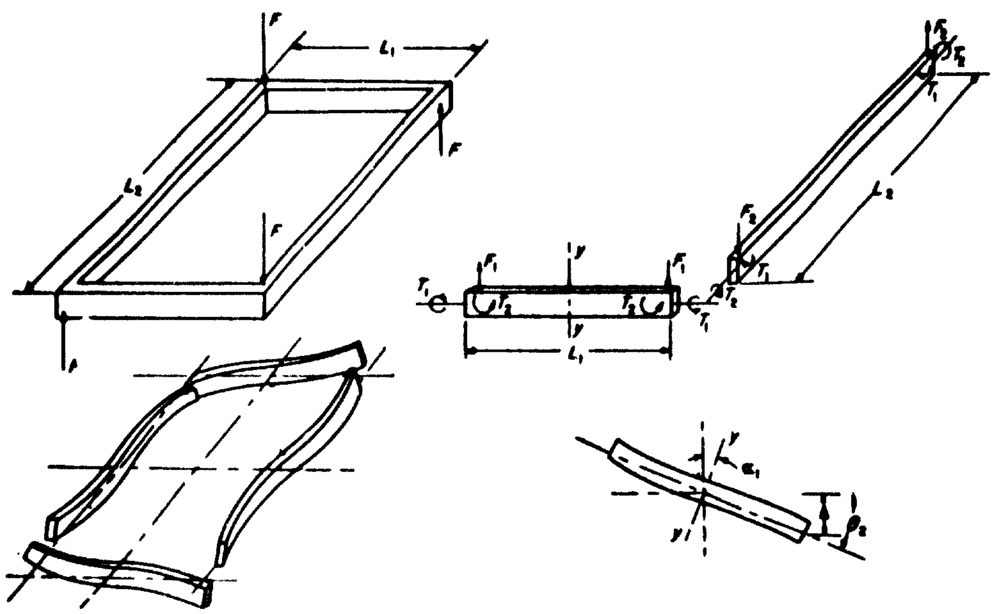

Fig.18 The frame showing the force and moment system acting on its sides.

Side L_1 can be treated as two cantilevers of length $\frac{L_1}{2}$ acted on by force F_1 at the free end and by bending moment T_2 which is equal to the torsional moment acting on side L_2. The free end of the cantilever deflects by an amount δ relative to its support. A vertical plane through the root of the cantilever in the unloaded state will not remain vertical but will rotate through an angle a due to the vertical displacements of the sides L_1. The total deflection of the cantilever end is, therefore, $(\delta_1 + \frac{L_1}{2} a_1) = \Delta$, and the slope at the end is (a_1 + slope of cantilever relative to its support).

Plain and transverse-braced frames

The bending moment acting at distance x from the free end of the cantilever $\frac{L_1}{2}$ is $(F_1 x - T_2)$

$$\therefore \quad EI_1 \frac{d^2y}{dx^2} = F_1 x - T_2$$

$$EI_1 \frac{dy}{dx} = \frac{F_1 x^2}{2} - T_2 x + A \qquad \ldots\ldots (8.1)$$

$$\frac{dy}{dx} = 0 \text{ for } x = \frac{L_1}{2} \quad \therefore \quad A = \frac{T_2 L_1}{2} - \frac{F_1 L_1^2}{8}$$

$$\therefore \quad EI_1(y) = \frac{F_1 x^3}{6} - \frac{T_2 x^2}{2} - \frac{F_1 L_1^2}{8} - \frac{T_2 L_1}{2} x + B$$

$$y = 0 \text{ at } x = \frac{L_1}{2} \quad \therefore \quad B = \frac{F_1 L_1^3}{24} - \frac{T_2 L_1^2}{8}$$

$$\therefore y = \frac{1}{EI_1}\left[\frac{F_1 x^3}{6} - \frac{T_2 x^2}{2} - \left(\frac{F_1 L_1^2}{8} + \frac{T_2 L_1}{2}\right) x + \frac{F_1 L_1^3}{24} \quad \frac{T_2 L_1^2}{8}\right]$$

$$\text{For } x = 0, \; y = \delta_1 = \frac{1}{EI_1}\left(\frac{F_1 L_1^3}{24} - \frac{T_2 L_1^2}{8}\right) \qquad \ldots\ldots (8.2)$$

Also $\Delta = \delta_1 + a_1 \frac{L_1}{2}$ $\therefore a_1 = \frac{2}{L_1}(\Delta - \delta_1)$ ..... (8.3)

and θ_2 = angle of twist of side L_2

$$= a_1 \left[\frac{dy}{dx}\right]_{x = 0}$$

$$\therefore \theta_2 = a_1 + \frac{1}{EI_1}\left[\frac{T_2L_1}{2} - \frac{F_1L_1^2}{8}\right] \quad \text{.....(8.4)}$$

From the torque equation, $T_2 = \frac{2GR_2\theta_2}{L_2}$ since each half of side L_2 twists by θ_2 relative to the centre section.

$$\therefore \theta_2 = \frac{T_2L_2}{2GR_2} = \frac{2}{L_1}\left[\Delta - \frac{1}{EI_1}\left(\frac{F_1L_1^3}{24} - \frac{T_2L_1^2}{8}\right)\right]$$

$$+ \frac{1}{EI_1}\left[\frac{T_2L_1}{2} - \frac{F_1L_1^2}{8}\right]$$

$$\therefore \Delta = \frac{T_2L_1L_2}{4GR_2} + \frac{1}{EI_1}\left[\frac{5F_1L_1^3}{48} - \frac{3}{8}T_2L_1^2\right]$$

But $T_2 = \frac{F_1L_1}{2}$ $\therefore \frac{F_1}{\Delta} = \frac{1}{\frac{L_1^2L_2}{8GR_2} - \frac{L_1^3}{12EI_1}} = k_1$

Similarly $\frac{F_2}{\Delta} = \frac{1}{\frac{L_1L_2^2}{8GR_1} - \frac{L_2^3}{12EI_2}} = k_2$

$\therefore \frac{F_1+F_2}{\Delta} = \frac{F}{\Delta} = k_1 + k_2$ = linear stiffness of each corner

Torsional stiffness $\frac{T}{2\theta} = \frac{(k_1+k_2)\,\Delta L_1}{4\Delta/L_1} = \frac{(k_1+k_2)L_1^2}{4}$ .....(8.5)

For a frame with transverse bracing, the torsional stiffness is $\frac{1}{n}$ times that derived by Eqn 8.5 approximately. More accurately, the effect of each intermediate transverse member can be divided equally between adjacent cells. The stiffness of a frame with n cells is given by

$$\frac{1}{n}\left[K_1 + (n-1)K_2\right]$$

where K_1 and K_2 are the torsional stiffnesses of cells with transverse members of torsional resistances R_1 and $\frac{R_1}{2}$ and second moments of area I_1 and $\frac{I_1}{2}$ respectively.

Diagonal-braced frames

In the case of the double diagonal-braced frame, where the diagonals are at 90° approximately, twisting of the diagonals can be ignored, and each diagonal of length L_3 can be treated as two cantilevers of length $\frac{L_3}{2}$. In this case, the free ends are deflected in the same direction, so the root section does not change its position under load.

$$\therefore \quad \frac{F_3}{\Delta} = \frac{24EI_3}{L_3^{\,3}} = k_3$$

is the stiffness of the diagonals, and the total stiffness is $k + k_3$, where k is the stiffness of the outline frame. For 45° bracing

$$L_3 = \sqrt{2}L_1$$

$$\therefore \quad k_3 = 8.49\,\frac{EI_3}{L_1^{\,3}}$$

and torsional stiffness is

$$k + \frac{k_3}{n}\,\frac{L_1^{\,2}}{4} \qquad \ldots\ldots (8.6)$$

Tests on model frameworks showed that double-diagonal bracing gave an increase in stiffness of 830% over the plain frame for a 94% increase in material, corresponding figures for the single diagonal frame were 357% increase in stiffness for 47% more material, and for the cross-braced frame, 47% increase in stiffness for 33% more material. Compared with the cross-braced system, the double diagonal frame yielded a 558% stiffness gain for a 46% increase in material, the corresponding figures for the single-diagonal frame being 220 and 10.3%.

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